CHAPTER 2

HVAC CONTROL SYSTEM EQUIPMENT, EQUIPMENT USES AND HVAC CONTROL LOOPS

1. GENERAL. The design of HVAC control systems is implemented by defining the operating modes of the HVAC equipment, defining the control loops required, and selecting the control system equipment to be used in the loop. The process of selecting the control system equipment includes calculations by the designer to specify the flow capacity of control devices, the physical size of control devices and the electric service required. This chapter describes the operating modes, process variables, control modes, control system devices and their features, control system equipment applications, and inter-connection of control devices. This chapter provides criteria and guidance for selecting and sizing control devices.

2. CONTROL SYSTEM OPERATING MODES AND PROCESS VARIABLES.

a. Control system operating modes. Control systems start and stop the HVAC system equipment according to a time schedule, and at specific outside air temperatures and specific indoor temperatures. In addition, the control systems operate the HVAC systems in the following modes of operation:

(1) Occupied mode is initiated automatically to allow HVAC systems to start in sufficient time to bring the space to the proper temperatures at the start of occupancy.

(2) Ventilation delay mode is initiated automatically to prevent the use of outside air when the unit is started prior to occupancy, to cool down or warm up the area served.

(3) Unoccupied mode is initiated automatically to prevent unnecessary operation of HVAC-system equipment during periods of non-occupancy except for special purposes such as operation to maintain minimum space temperatures for freeze protection.

(4) Heating or cooling modes are initiated manually to provide either heating or cooling media to HVAC equipment.

b. Control system process variables. While the HVAC systems are in operation, the process variables commonly sensed and controlled by HVAC control systems are:

(1) Temperature.
(2) Relative humidity.
(3) Static pressure of air.
(4) Differential pressure of air.
(5) Air flow rate.

c. Constraints on process variables by operating modes. The constraints placed on the control of HVAC process variables by the operating modes are:

(1) Cooling and humidification are shut off during the unoccupied mode.
(2) Outside air is not supplied to the space during the unoccupied and ventilation delay modes.

d. Modulating control. The amount of heat delivered to a space (or removed from a space) from certain types of HVAC equipment is regulated by varying the heat exchanger capacity from zero to one-hundred percent in response to the variation of a continuous, gradual input signal. This is called modulating control. Heat exchanger control valves, mixing dampers, fan inlet vanes, variable speed drives, and humidifier valves are examples of HVAC equipment that are controlled by modulating control.

e. Two-position control. The amount of heat delivered to a space from certain types of HVAC equipment is controlled by turning the equipment on and by shutting the equipment off. This type of control is also called on-off control. Examples of 2-position control are the starting and stopping of the fans of unit heaters and fan coil units by room thermostats to maintain space temperature, and the opening and closing of shutoff dampers when fans are started and stopped.

3. CONTROL SYSTEM EQUIPMENT.

a. Control valves.

(1) Control valves are used to regulate the flow of fluids in piping systems by compressing and releasing a valve spring to move a valve closure disk or plug toward or away from the closure seat of a flow port. The valves are used both in modulating and in 2-position control applications.

(2) Examples of the use of modulating control valves are:

(a) Heating and cooling coil control valves.
(b) Converter steam control valves.
(c) Humidifier control valves.
(d) Perimeter radiation system zone valves.

(3) Examples of the use of 2-position control valves are:

(a) Dual-temperature water system changeover valves.
(b) Shutoff valves used in fan coil unit coils.

(4) Control valves are classified according to their flow regulating body patterns. A 2-way valve restricts fluid flow in one direction, because it has one inlet and one outlet; a 3-way valve restricts flow in two directions. The designer will use 2-way control valves for controlling the following types of HVAC equipment:

(a) Convertors.
(b) Radiators.
(c) Coils served by variable volume pumping systems.
(d) Steam coils.

(5) A 3-way mixing valve has two inlets and one outlet, and a 3-way bypass (diverting) valve has one inlet and two outlets. For the systems shown in this manual, the 3-way mixing valve is used in both flow mixing and flow diverting type applications, except that a 3-way bypass valve is used as a dual-temperature changeover control valve.

(6) In the flow mixing application, the 3-way valve is used to mix heated primary flow, from a boiler or a converter, with system return flow to produce system secondary supply, for the purpose of controlling temperature. When used on the return line from a coil, one of the 3-way valve's inlets is from the coil, and the other inlet is from the bypass around the coil. The designer may choose to use 3-way mixing valves in lieu of the 2-way valves shown in this manual for controlling the following types of HVAC equipment to prevent deadheading of pumps:

(a) Coils served by constant volume pumping systems.
(b) As means of pump pressure relief in variable volume pumping systems.
(c) As perimeter radiation zone valves.
(d) As diverting valves around boilers or cooling towers.

(7) The designer may choose to use either four 2-way valves or one 3-way mixing valve, and one 3-way bypass valve for 2-position flow control applications as dual-temperature system changeover valves.

(8) Control valves are classified according to the action of the valve spring in moving the disk or plug relative to the seat when the control signal or the power is removed. A 2-way valve that opens its flow port under this condition is called a normally open (NO) valve, and one that closes its flow port under this condition is called a normally closed (NC) valve. A 3-way mixing valve has both NC and NO inlet flow ports connected to a common (C) outlet flow port. A 3-way bypass valve has both NC and NO outlet flow ports connected to a C inlet flow port.

(9) The flow regulating characteristic of a valve is generally determined by the shape of a disk or plug that passes through the flow port. The flow regulating characteristics used for control systems covered by this manual are:

(a) Linear flow, in which the percent of valve travel equals the percent of maximum flow rate through the valve.

(b) Equal-percentage flow in which equal increments in the percentage of valve travel produce an equal-percentage change in flow rate from the previous flow rate, when a constant pressure drop is maintained.

(10) The applications of 3-way mixing valves covered by this manual require the use of valves with linear flow characteristics. The applications of 2-way valves covered by this manual require the use of valves with equal-percentage flow characteristics. This requirement results from the application of the valves as modulating fluid control devices. The equal-percentage flow characteristic matches the non-linear heat exchange characteristics of the HVAC equipment coils with a change in fluid flow that tends to linearize the heat exchange output of the coil with a linear signal to the control valve. The linear-flow characteristic is more suitable for mixing applications and for humidification.

(11) The purchase price of a control valve increases with its size. The installation cost of a control valve also increases with its size, because of:

(a) The change from screwed ends to flanged ends
(b) Because larger valves and their weight require more installation and handling labor.

(12) At a pipe size of 4 inches or larger, a type of rotary control valve (known as a butterfly valve) becomes economically suitable for HVAC control applications because of the combination of the price of the valve and the installation costs. The butterfly valve has a disk that rotates on a shaft and closes against a seat. The seat is concentric with the connected pipe. The butterfly valve can have flow control characteristics similar to equal percentage when used with an appropriate actuator and positioner. In three-way applications of valves for 4-inch pipe size and larger, the designer will show two valves on a common pipe tee, with separate actuators that will operate the two valves simultaneously. One of the valves will be NC, and one will be NO. The C connection can be either an inlet or an outlet. This allows the combination of two valves and a pipe tee to function as a 3-way mixing valve or a 3-way bypass valve. Figure 2-1 shows butterfly valves used in 3-way mixing and 3-way bypass arrangements on a common pipe tee.

Figure 2-1. Two butterfly valves on a common pipe tee.

b. Control dampers.

(1) Dampers are used to regulate the flow of air in ductwork in both modulating and 2-position control applications.

(2) Examples of the use of modulating dampers are:

(a) Air plenum temperature control by mixing outside air and return air.
(b) Space temperature control by mixing warm air and cool air.
(c) Space temperature control by varying the flow of cool air.

(3) Examples of the use of 2-position dampers are:

(a) Closing outside air dampers or building exhaust dampers when fans are stopped.
(b) Isolating sections of ductwork for smoke control purposes.

(4) Dampers are classified by the action of their blades, which connect to a common shaft that is rotated to open or to close the damper. Opposed blade dampers provide better flow characteristic in throttling applications. A throttling application is one where the damper is installed in series with the path of flow and the damper is used to add pressure drop to reduce air flow. Parallel blade dampers are used to provide better flow characteristics in mixing applications. A mixing application is one where more than one flow path exists in parallel. Usually, two or more dampers are installed in parallel to each other and the dampers divert flow rather than increase total system pressure drop.

(5) The control action of dampers (NC or NO) depends on the direction of their blade rotation caused by the spring return stroke of an actuator connected to the damper's drive shaft, when the control signal or power is removed.

(6) When a control system application requires that a damper be open prior to the start of a fan, an adjustable switch is connected to the damper; this device is called an end switch or limit switch. The end switch operates a set of contacts in the fan starter control circuit when the damper is fully open, to allow the fan to start; the end switch opens the circuit to prevent the fan from continuing to operate if the damper begins to close.

c. Actuators.

(1) Actuators are used to operate valves and dampers. Pneumatic actuators are powered by air pressure, and are controlled directly by a pneumatic control signal and indirectly by an electric or electronic signal. An electro-pneumatic device converts an electric or electronic signal to a pneumatic signal to stroke the actuator. Electric and electronic actuators are electrically powered and are controlled directly from an electric or electronic signal to stroke the actuator. While all pneumatic actuators have a spring-return feature, some electric/ electronic actuators are not equipped with aspring to move the valve or damper to a fail-safe position upon loss of power or control signal.

(2) Modulating control of actuators requires either the use of a 4 to 20 milliampere control signal directly to an electronic actuator or the conversion of the signal to a pneumatic control signal of 21 to 103 kPa (3 to 15 psig). The pneumatic signal can be directly or inversely proportional to the electronic signal. The signal conversion values are shown in figure 2-2.

Figure 2-2. Conversion of an electronic signal to a pneumatic signal.

(3) Two-position control of electric actuators requires the closing and opening of a contact to operate an electric actuator. Two-position control of pneumatic actuators requires an electric/pneumatic device to pass 140 kPa (20 psig) main air to the actuator, or to exhaust air from the actuator.

(4) Sequencing occurs when actuators are modulated from a common signal by using a portion of the 4 to 20 milliampere signal or the converted 21 to 103 kPa (3 to 15 psig) signal. The actuator stroke is adjusted to move its connected valve or damper from fully closed to fully open over the assigned portion of the common control signal. Deadbands between the movement of valves and dampers are achieved by

assigning a portion of the common control signal as a deadband. Each actuator is adjusted so that its full stroke occurs on either side of the deadband limits outside of the deadband. Examples of the use of sequencing with a deadband are:

(a) Sequencing of heating and cooling with a deadband between heating and cooling.

(b) Sequencing of heating and outside ventilation air beyond the required minimum quantity with a deadband between heating and increased ventilation.

(5) Actuators are modulated in parallel by assigning the identical portion of the control signal to each actuator for its full stroke. Modulation in parallel occurs in air stream mixing applications such as:

(a) Modulation of outside air, return air and relief air dampers for free cooling.

(b) Modulation of multizone hot deck and cold deck dampers in parallel.

d. Current-to-pneumatic transducers. The modulating device for converting a current control signal to a pneumatic control signal is a current-to-pneumatic transducer (IP). A 140 kPa (20 psig) main air supply to the IP is the source that develops a 21 - 103 kPa (3 - 15 psig) output signal in a scaled relationship to a 4 - 20 milliampere input signal.

e. Solenoid operated pneumatic valves. The 2-position device for converting an electric contact closure signal to a pneumatic signal is the solenoid operated pneumatic valve (EP). The EP is a 3-way valve that connects the normally closed and common ports when the solenoid coil is energized and connects the normally open and common ports when the solenoid coil is de-energized. The EP is used to switch 140 kPa (20 psig) main air to the actuators and to exhaust air from the actuators.

f. Positive Positioners.

(1) All modulating control applications of pneumatic actuators require that the actuator be equipped with a positive positioner (PP). A main air supply is the source of its operating power. The device throttles main air as required to stroke the actuator to the position dictated by the pneumatic control signal. However, the positive positioner can exert pressure higher than that of the pneumatic control signal and thus can maintain the required position against the opposing force of the HVAC system pressure. Piping system pressures tend to compress the air in the diaphragm chamber of the valve actuator. The compression causes a shift in the actual operating ranges of the valves. The positive positioner has an adjustable pneumatic signal start point for the stroke of the actuator and an adjustable pressure span for the full stroke of the actuator. The stroke is proportional to the pneumatic control signal.

(2) Non-modulating (two-position) control applications where pneumatic actuators are used do not require positive positioners.

(3) Simultaneous heating and cooling can occur when pneumatic actuators are used, even though the spring operating ranges are selected without an overlap. The results of this phenomenon are shown in figure 2-3. Because of this phenomenon, sequencing applications for HVAC systems must have positive positioners on pneumatic valves and damper actuators, to maintain deadbands between actuator operating ranges. A control system with positive positioners is illustrated in figure 2-4. When sequencing actuators from a common control signal, the simultaneous use of heating and cooling can accidentally occur if:

(a) Heating and cooling valve operating ranges overlap.
(b) Heating valve and ventilation damper operating ranges overlap.
(c) Heating valve and cooling air damper operating ranges overlap.

Figure 2-3. Simultaneous heating and cooling with pneumatic actuators without positive positioners.

Figure 2-4. Control system with positive positioners to avoid simultaneous heating and cooling.

g. The choice between pneumatic and electric actuators. All terminal unit control systems will have electric or electronic actuators. For all other control system applications, the designer will make an estimate of the total cost of actuators required for all control systems in the project. The designer will take into account the cost of multiple actuators on large dampers and the cost of larger actuators required for higher torques to operate large valves. The total installed cost estimate of pneumatic actuators will include:

(1) The actuators.
(2) The IPs.
(3) Tubing.
(4) Local indicators.
(5) The cost of the compressed air system.

The total installed cost estimate of electric actuators will include consideration of:

(1) The actuators.
(2) Wiring.
(3) Loop driving circuits as explained in this manual.
(4) Power transformers (24 VAC)

h. Existing compressed air source. If sufficient air is available from an existing temperature control compressed air system, it may be used as the air source for additional control systems.

i. Life cycle cost. After the installed cost estimates are prepared, a life cycle cost estimate will determine the choice between pneumatic and electric actuators. Some manufacturers' catalogs provide guidelines to assist in estimating the cost benefits of using electric versus pneumatic actuators.

j. Sequencing actuators. The actuators that control valves and dampers are sequenced when HVAC applications require that the process variables be sensed at a common location and controlled from a common modulating signal. The objective of sequencing is to avoid energy waste by preventing the following opposing processes from acting simultaneously:

(1) Heating and cooling.
(2) Humidification and dehumidification.

k. Design requirement in regard to actuator sequencing ranges. The designer will show the actuator sequencing ranges in the equipment schedule when standard control signals apply.

(1) Pneumatic actuators are sequenced by connecting the signal input connections of the actuators' positive positioners to the same pneumatic control signal and adjusting the positioners' starting points and spans to achieve the required sequence. For example, two valves can be operated in sequence if their positive-positioner spans are set at 28 kPa (4 psig) and their starting points are set at 21 kPA (3 psig) and 62 kPa (9 psig) respectively. This results in ranges of valve full-stroke operation of 21 to 48 kPa (3 to 7 psig) and 62 to 90 kPa (9 to 13 psig), with a 14 kPa (2 psig) deadband between the ranges of operation.

(2) Some electric actuators have starting points and span adjustments similar to those of the pneumatic actuator's positive positioner. This is sometimes an optional feature, and must be specified if required for sequencing. In this case, the starting points and spans are adjusted in milliampere values. When electric actuators are sequenced, the modulating control circuit will be designed within a 600 ohm limitation.

l. Multiple actuators connected to the same control damper. When the operating torque requirement for an HVAC system damper exceeds the output torque of a single actuator, additional actuators are connected together to operate in parallel to control the damper. The designer is not required to show multiple actuators connected to the same damper on the schematic. The vendor has the information necessary in the contract specification to apply multiple actuators when required.

m. Design of modulating control circuits within a 600 ohm circuit impedance limitation.

(1) The output of an HVAC system controller is connected in series to actuators external to the HVAC control panel, and also to other devices in the HVAC control panel in a direct current series circuit. The number of devices varies with the complexity of the control sequence, and the impedance of each connected device is additive as a resistance in the circuit. The amount of output circuit impedance that a controller will tolerate is product specific. The limitation of 600 ohms in the output circuit design is needed to permit the controllers of several manufacturers to function in the same circuit. The limitation permits standardization in the design and permits substitution of one manufacturer's controller for that of another during maintenance of the system.

(2) If a modulating control circuit is designed to use electric or electronic actuators, the impedance can exceed the 600 ohm limitation if:

(a) Multiple actuators are required for the same damper.

(b) More than one damper is modulated from the same control signal, such as in the case of modulating outside air, return air, and relief air dampers.

(c) Multiple control system devices located within the HVAC control panel are necessary to achieve the sequence of control.

(3) Individual control system devices typically add 250 ohms impedance to a series circuit. This 250 ohm impedance value comes from a dropping resistor in the device that is used to convert the 4 - 20 milliampere current signal at 24 volts dc to a 1 - 5 volt signal used by the device's internal circuitry.

(4) Figure 2-5 shows methods for designing circuits within the 600 ohm limitation. The figure shows the following examples:

(a) The limitation exceeded by connecting 750 ohms in series.

(b) Limiting the control circuit connection to a single actuator.
(c) The circuit designed within the limitation by the use of an additional control circuit device.

Figure 2-5. Modulating control circuits impedance limitation.

(5) In the first example, shown in the upper part of the figure 2-5, control devices 1, 2 and 3 can be actuators or devices in the HVAC control panel. In the second example, the controller is connected to actuator 1, and actuator 2 is operated by an auxiliary actuator driver (AAD) circuit on actuator 1; actuator 2 can operate another actuator by its AAD. (Note that the functionality provided by the AAD is usually a built-in feature of the actuator and the AAD is not necessarily a separate device.) In the third example, the same control devices are connected to a loop driver (LD). Control device 1 and the loop driver are connected to the controller. The modulating circuit from the controller is limited to 500 ohms, consisting of 250 ohms for control device 1 and 250 ohms for the input to LD. Control devices 2 and 3 add a total of 500 ohms to the output circuit of LD. The output signal of LD varies in a 1:1 ratio with its input signal.

(6) Some of the control devices necessary to implement the control sequence have an input impedance of 250 ohms, and their output circuits can accept from 800 ohms to 1000 ohms of impedance. The amount of allowable impedance in their output circuits is product specific. Any control device whose modulating control output circuit has greater impedance loading capability than the impedance of its modulating control input circuit can function as a loop driver, in addition to performing a specific control sequence function. This output driving capability is found in most modulating control devices.

(7) When the control system requires more than one damper with electric or electronic actuators to be modulated by a control circuit, the designer will show the signal (on the schematic) connected to one of the damper actuators. The AAD circuit of that actuator will be shown as connected to drive a separate actuator on another damper, which, in turn, can drive another actuator on still another damper.

(8) When a modulating control circuit must drive multiple panel mounted control devices, the designer will show on the schematic:

(a) Not more than 2 devices (such as IPs) connected to that circuit, unless one of the devices is a control device that accepts a modulating input signal and produces a modulating output signal.

(b) Not more than two panel mounted control devices connected to the modulating output of a panel mounted control device.

(9) The schematic is not intended to show the physical connections to the devices, but rather to show the relationship of the necessary control devices in the control loop.

n. Transmitters. Variables such as temperature, pressure, and relative humidity are sensed by means of elements that are connected to the control loops via transmitters. The output signal of the transmitter is the standard 4 to 20 milliampere dc signal, which is factory calibrated for zero point and span relative to the input resistance value of the sensing element. The transmitters are 2-wire, loop-powered (i.e., powered by the control panel power supply) devices that connect in a series circuit with the controller input. The impedance limitation of the circuit in which the transmitter can function is product specific. A typical value is 700 ohms at 24 volts dc.

o. Single loop digital controller.

(1) As shown for the standard control systems, single loop controllers are used for essentially all systems other than simple unitary systems and terminal units that are controlled directly from room or zone thermostats. In all applications where it is used, the controller will be mounted in a HVAC control panel. The controller mounting dimensions will conform to a standard panel cutout requirement. The controller will be used for the following applications:

(a) As a controller for maintaining temperature, relative humidity, static pressure, and/or airflow setpoints.

(b) As an economizer mode switchover controller that determines whether outside air is suitable for cooling.

(c) As an outside air temperature controller for scheduling hydronic heating supply temperature and for starting and stopping pumps.

(2) The controller will be a microprocessor-based device with manually configurable control features resident in solid state electronic memory components. Manual access to the features of the controller will be through a keypad and an alphanumeric indicator on the face of the controller. The controller will have standard features that will allow it to serve all functions prescribed in its application.

Standard features will allow the replacement of any controller by a spare not necessarily of the same manufacturer.

(3) The single-loop digital controller will have the following inputs and outputs:

(a) A process variable analog input (PV).

(b) A remote setpoint analog input for control point adjustment (CPA).

(c) An analog output (OUT).

(d) A process variable actuated contact closure output (PV contact).

(e) A contact output that responds to the difference between PV and CPA analog inputs (DEV contact).

(f) An analog output which is identical to the process variable input (PV Retransmission).

(4) Some controller features are configurable by manual input, and others are selectable by setting switches or jumper wires. The configuration methods vary with the specific manufacturer and model. The controller will have the selection of features for its inputs and outputs as shown in table 2-1.

TABLE 2-1 CONTROLLER FEATURES

Input	Output	Feature
PV		Scalable to the range of the input transmitter.
CPA		Adjustable bias of setpoint relative to the PV input range.
CPA		Adjustable ratio of setpoint relative to the PV input range.
CPA		Scalable to the range of the CPA input signal.
CPA		High and low limits of setpoint adjustment.
	OUT	Selection of direct acting or reverse acting.
	OUT	Adjustable high and low limits.
	OUT	Adjustable bias (Manual reset).
	OUT	Selection of manual or automatic control.
	OUT	Selection of proportional (P), proportional plus integral (PI), and proportional plus integral plus derivative control modes (PID).
	OUT	Anti-reset windup.
	OUT	Selection of manual tuning or operator initiated self tuning.
	OUT	Manual reset feature for use when the controller is configured from porportional mode only and is manually tuned.
	PV Contact	Adjustable contact setpoint.

PV Contact	Adjustable hysteresis or deadband.
DEV Contact	Adjustable contact setpoint.
DEV Contact	Adjustable hysteresis or deadband.

(5) The output control modes that can be used in combination are:

(a) Proportional mode, which varies the output proportionally to the error between the PV input value and the controller setpoint.

(b) Integral mode, which modifies the output signal as a time related function of the error between the PV input value and the controller setpoint.

(c) Derivative mode, which modifies the output signal as function of the rate of change of the error between the PV input value and the controller setpoint.

(6) Each of these control modes has an assignable constant, which is adjusted in the process of tuning the controller.

(7) The integral mode will have a feature that automatically stops the integration of the error signal when the controller output signal reaches its minimum or maximum value. This feature is known as anti-reset windup. If the process being controlled does not respond to the controller output for a period of time (such as an overnight shutdown), continuing the integration of the error signal during that period would result in the controller failing to respond to control the process immediately on startup. Without the anti-reset windup feature, the controller would reverse the integration process on startup, but the reversal would require a time period equal to the time period of the integration during shutdown.

(8) The controller will have the following selectable modulating control functions:

(a) Self tuning of control mode constants.

(b) Manual tuning of control mode constants.

(9) Self tuning allows the controller to select the optimal combination of proportional, integral, and derivative control mode constants. The controller continues to use these selected constants until the self-tuning control function is again selected. The Guide Specification requires that the self-tuning process be operator initiated. Applications where self tuning is used are as follows:

(a) Mixed air temperature control.

(b) Heating coil temperature control for multizone hot deck and dual-duct hot duct applications.

(c) Modulating valve and modulating damper preheat coil control.

(d) Cooling coil discharge temperature control for multizone cold-deck, dual-duct cold-duct, and VAV system discharge temperature.

(e) Relative humidity control.

(f) Supply duct static pressure control.

(g) Return fan volume control.

(h) Hydronic heating supply temperature control.

(i) Hydronic-heating space-temperature control.

(10) If integral and derivative control modes are not appropriate for an application, the controller is manually tuned for proportional mode control, and the integral and derivative mode constants are set to zero. Applications where manual tuning is used are as follows:

(a) Relative humidity high limit control.

(b) Outdoor air scheduling of hydronic heating supply temperatures or hot deck/ heating coil discharge air temperatures.

(c) Space temperature control of single-zone HVAC units and heating and ventilating units.

(11) When the single-loop digital controller is manually tuned to operate as a temperature controller in the proportional mode, the proportional band setting is determined by the use of equation 2-1.

$$PB = \frac{(TR \times 100)}{T_s} \qquad \text{(eq. 2-1)}$$

Where:

PB = Proportional band constant (percent).
TR = Throttling range or the portion of the transmitter span required for full-scale controller output change (deg. C (deg. F)).
T_s = The temperature span of the transmitter (degrees C (degrees F)).

(12) If the controller output is to change from 4 to 20 milliamperes over the range of minus 18 to plus 16 degrees C (0 to 60 degrees F) and the transmitter range is in the range of minus 35 to plus 55 degrees C (minus 30 to plus 130 degrees F) (a span of 90 degrees C (160 degrees F)), the use of equation 2-1 would result in: PB = 37.5 percent.

(13) When a single-loop controller is configured for proportional control mode only, the output of the controller must be set to match the value required for the application when the PV input value coincides with the controller setpoint. This is accomplished by configuring the manual reset setting as a percent of controller output. The effect of this configuration parameter is to shift the controller output throttling range with respect to the controller setpoint. As an example, refer to figure 2-6 which shows the setpoint relative to the controller throttling range for manual reset settings of 25, 50 and 75 percent for a direct-acting controller.

Figure 2-6. Manual reset feature.

(14) When used as an economizer switching controller, the functions of the controller inputs and outputs will be as follows:

(a) A return air temperature transmitter connected to the PV input.

(b) An outside air temperature transmitter connected to the CPA input.

(c) An output contact configured as PV and acting as a switch.

(d) An output contact configured as DEV and acting as a switch.

(e) There are exceptions to the above listed connections (such as when an economizer is used in a multizone system.

(15) As the outside air temperature changes, the potential for using outside air for cooling is affected. When the outside air temperature is lower by a specified number of degrees than the return air temperature, outside air can be used for cooling; when the reverse is true, outside air must be at minimum quantity. The PV contact setpoint prevents the use of outside air beyond minimum quantity until the return air temperature rises to its setpoint. When the return air temperature rises to the PV contact setpoint, the HVAC system is no longer experiencing a heating load. The DEV contact setpoint allows the use of outside air until the outside air temperature approaches the return air temperature. The DEV contact then puts the dampers under control of a minimum-position switch. A minimum-position switch is a device whose control output is manually modulated to an actuator that then remains fixed until reset manually. Both conditions must be met (PV and DEV contacts closed) for the use of outside air beyond minimum quantity.

p. Function Modules.

(1) There are control functions required with less frequency in HVAC control applications than those included in the prescribed version of the single-loop controller. Control devices to perform specific control functions not available in the single-loop digital controller are called function modules in this manual. Function modules will be located in the HVAC control panel, except as noted. Function modules accept contact, analog, or gradual manual adjustment input signals to provide:

(a) Signal selection,

(b) Signal inverting,

(c) Contact transfer output from analog signal input,
(d) Contact transfer output from the comparison of two analog input signals, and

(e) Generation of constant analog output signals.

(2) A minimum-position switch is a manually adjustable modulating output device used to hold an outside air damper open to admit minimum ventilation air. This same device is used as a temperature setpoint selector. In this application, it allows manual adjustment of the CPA input signal to a single-loop digital controller. In the case of certain single-zone HVAC systems, this device is wall-mounted and accessible to the occupant for adjustment of the space temperature controller setpoint.

(3) A signal invertor is a modulating input and modulating output device used to reverse the direction of its input. The signal reversal is required when spring return position of a valve or damper actuator has been chosen to operate with a direct acting signal, and when an actuator with which it must be sequenced has been chosen to operate with a reverse acting signal. For example, a chilled water coil is used for dehumidification and cooling. The chilled water valve is chosen as NC because it is to be sequenced with a NO heating coil valve, and is to close when the control signal is removed during fan shutdown and during the unoccupied mode. The direct acting temperature control signal is correct for this combination. The humidifier control valve is also chosen as NC because it also is to close when the control signal is removed on fan shutdown and during the unoccupied mode. The humidity control signal must be reverse acting when it operates the humidifier valve, and must be direct acting when it operates the cooling coil valve. A signal invertor is used to reverse the signal direction of the humidity controller signal to the cooling coil valve.

(4) A signal selector is a device with multiple modulating inputs and a modulating output. This device is used to select the highest or the lowest of its input signals as its output.

(5) A sequencer is a device with a modulating input and one or more contact outputs which operate in sequence and is used in applications that require the operation of stages in refrigeration control from a modulating control signal. A deadband surrounding each of the sequencer setpoints prevents all stages of refrigeration from starting simultaneously when the HVAC system starts. The contact opens instantaneously on power failure (or on signal failure) to the sequencer.

(6) A loop-driver module is used where required to allow modulating control circuits to be designed within a 600 ohm impedance limitation. A loop-driver module can be any modulating input or modulating output device used for this purpose alone or while performing an additional control loop function. Signal selectors and signal invertors strategically located in control loops often perform such dual service.

q. Relays, including time-delay relays. All relays, including time-delay relays, will be 2-pole, double-throw devices; they are used for control system interlocking functions and will be located in the system's HVAC control panel.

r. Time clocks.

(1) A time clock will be used to control the timing of the modes of operation of an HVAC control system when the control system is not interfaced with EMCS. When a time clock is used, it will be located in a HVAC control panel. The modes of operation are occupied, unoccupied and ventilation delay.

(2) The time clock will be a device that accepts a time schedule by manual input through a keypad and an alphanumeric display. The time clock features will be:

(a) Four independent time-controlled contacts.

(b) A program of 4 "on" events and 4 "off" events for each contact.

(c) 365 day schedule.

(d) Twelve selectable holidays.

(e) Standard-time and daylight-time adjustments.

(f) Timed override of programs.

(g) Battery backup of memory.

(3) When used to time the modes of operation of air handling systems, one contact of the clock will be used for occupied and unoccupied timing; the second contact will be used for ventilation delay mode timing. For other applications, the contacts may be used as convenient to the design.

4. CONTROL LOOPS. A control loop performs three distinct functions: sensing of a variable as the input to a controller; decision making or control based on the value of the input; and output or actuation as a result of control. Figure 2-7 illustrates a simple control loop. The input signal is a continuous analog of the process, and the controller either continuously sees the input or continually scans it. The controller changes its output as required by changes in its input.

Figure 2-7. Control loop.

5. OPEN CONTROL LOOPS. When a control loop senses a variable, makes a control decision, and sends an output signal to a control device without receiving input information related to the results of its control action, the control loop is said to be an open loop. There are some open-loop control applications used in HVAC control, such as:

(1) Operation of pumps above or below a certain outside temperature.

(2) Automatic stopping of HVAC systems based on outside air temperature.

(3) Scheduling of hydronic heating supply temperatures based on outside temperature.

(4) Timing and time-delay operations.

Figure 2-8 illustrates an open control loop.

Figure 2-8. Open control loop.

6. CLOSED CONTROL LOOPS. When the controller changes its output decision based on updated input information, the control loop is said to be a closed loop. Most of the control loops used in HVAC control are closed loops. Control of coil air discharge temperatures is an example. The transmitter, connected to a temperature sensing element in the air stream passing through the coil, signals the temperature controller; the controller makes a decision as to whether to open or close the valve that allows water to flow through the coil; and an actuator operates the valve. The feedback in this example is the continuous input to the controller of a changing temperature signal from the coil air discharge temperature sensor and transmitter. The transmitter continuously updates the controller on temperature information from the sensor, and the controller modifies its output to control the valve. See figure 2-9 for an example of a closed control loop.

Figure 2-9. Closed control loop.

7. APPLICATION OF OPEN-LOOP CONTROL AND CLOSED-LOOP CONTROL TO HVAC SYSTEMS.

a. Open loops and closed loops in combination. Open loops and closed loops are used in combinations in some HVAC control-system applications. A perimeter hydronic heating system may have open-loop components to start and stop the pump and to schedule the supply water temperature based on outside air temperature. At the same time, it may have a closed loop for the control of the supply water temperature.

b. Closed loops in combination. There are some HVAC control applications that use two simultaneously acting, closed control loops to actuate the same device. An example, as shown in figure 2-10, is control of a duct humidifier. A space relative humidity transmitter is the primary input to a relative humidity controller for the humidifier valve in a closed loop (loop 1). A duct-mounted humidity transmitter downstream of the humidifier signals a high limit relative humidity controller, which provides a high limit closed-loop control function (loop 2), by overriding the primary controller loop 1 to shut off humidification by closing the valve if the relative humidity of the air stream rises to the loop-2 relative humidity high-limit setting.

Figure 2-10. Two loops controlling one device (humidity control with high limit).

8. TYPICAL CONTROL MODES.

a. Two-position control.

(1) Some HVAC equipment can be turned on and off as a method of temperature control. This type of HVAC equipment is not applied where temperature control between close limits is required.

(2) When a thermostat or other control device cycles equipment to maintain its setpoint the control mode is called two-position control. A thermostat used for two-position control opens and closes contacts for control rather than providing a modulating output signal. The contacts either open or close when the temperature is at the thermostat setpoint. The state of the contact reverses when the temperature changes in the proper direction. Such thermostat contacts usually either open on a temperature rise (in a heating application), or close on a temperature rise (in a cooling application). The temperature at which this happens depends on the switch temperature differential (hysteresis).

(3) An example of two-position control is unit heater control, in which a space thermostat turns on a unit heater when the space temperature drops to 18 degrees C (65 degrees F) and turns it off when the space temperature rises to 19 degrees C (67 degrees F). The thermostat is said to have a differential of 1 degree C (2 degrees F) and a setpoint of 18 degrees C (65 degrees F). This type of control can result in a slight undershoot below the lower end of the differential, and a slight overshoot above the higher end of the differential.

(4) The thermostat may turn off the unit at 19 degrees C (67 degrees F), but the heating load may decrease due to increasing outside air temperatures. In this event, water circulating through the unit coil, which will then be acting as a radiator with the fan off, may raise the temperature in the space slightly above 19 degrees C (67 degrees F) as an apparent overshoot. Even though the unit turns on when the space temperature drops to 18 degrees C (65 degrees F), the space temperature may fall slightly below 18 degrees C (65 degrees F) after the unit starts. This depends on the heating load at the time and on the heating capacity of the unit. If the heating load decreases, the temperature may subsequently rise to 18.5 degrees C (66 degrees F) and stay at that temperature for a considerable time, while the fan continues to run. A similar situation can happen after the unit heater shuts off at 19 degrees C (67 degrees F) and the space temperature drops to 18.5 degrees C (66 degrees F). Consequently, the space temperature may be 18.5 degrees C (66 degrees F) with the unit heater fan either running or stopped. A graphic representation of two-position control is shown in figure 2-11.

Figure 2-11. Two-position control.

b. Modulating Control. A simple control loop is shown in figure 2-12 as it would be applied to heat outside air for ventilation using a pneumatic valve actuator rather than an electric or electronic valve actuator. The controller operates an IP in response to the signal of the temperature sensing element in the air duct, downstream of the coil, via a transmitter. The IP pneumatic output signal modulates the positioner on the pneumatic valve actuator. The positive-positioner output throttles main air to the actuator, which moves the valve stem. This example is used to explain two modes of modulating control that are applicable to the control of valves, dampers, inlet vanes, and other devices. The modes applicable to most HVAC control applications are:

(1) Proportional mode (P).

(2) Proportional plus integral mode (PI).

Figure 2-12. Simple control loop applied to outside air heating.

c. Proportional mode (P). The most common control mode in HVAC control is proportional mode.

(1) Proportional mode is used for the following applications:

(a) As a method of scaling an outside air temperature signal to schedule water temperatures for heating.

(b) As a method of space temperature control for single-zone air handling units.

(c) As a method of controlling terminal units that can be modulated.

(2) Figure 2-13 shows the kind of control that would be expected if the controller in figure 2-12 were configured for the proportional control mode. The controller modulates its output signal in proportion to variations of the input signal. For example, the controller, operating through the IP, sends a 21 kPa (3 psig) air signal to the normally open preheat coil valve when the discharge temperature is 6 degrees C (43 degrees F), and it sends a 103 kPa (15 psig) signal when the discharge temperature is 8 degrees C (47 degrees F). The 21 kPa (3 psig) signal completely opens the valve to heating, and the 103 kPa (15 psig) signal completely closes the valve to heating. The controller/IP combination has a proportional sensitivity of 37.2 kPa per degree C (3 psig per degree F), and throttles the valve over a range of 2 degrees C (4 degrees F). The setpoint of the controller is 7 degrees C (45 degrees F), but the temperature at which it is controlling is somewhere between 6 degrees C (43 degrees F) and 8 degrees C (47 degrees F). It takes a 2 degree C (4 degree F) change in controller output signal for the valve to go from full heating to no heating.

Figure 2-13. Proportional control mode.

(3) On a fall in outside air temperature, the heating load increases. The resultant drop in temperature at the sensor causes the transmitter to signal the controller to lower its output to the IP transducer. The IP transducer lowers the control air pressure to the normally open valve, allowing more heating medium to pass through the coil, thereby returning the discharge temperature almost to the temperature maintained before the fall in outside temperature. Increasing the controller's sensitivity would shrink the full heating/no heating temperature range; decreasing its sensitivity would increase that range.

(4) The phenomenon that prevents the proportional control mode from achieving its control point exactly at setpoint is called "offset due to load". This occurs when an equilibrium is reached between HVAC system output and the load imposed on the HVAC system. When this equilibrium occurs, the discharge temperature does not change. This means that the proportional mode controller does not change its output and cause a change in the HVAC system output. Conversely, without the change in HVAC system output to drive the temperature toward the controller setpoint, the controller output does not change. Therefore, at these equilibrium points, the controls do not bring the system back to setpoint.

(5) Figure 2-13 shows the relationship between coil discharge temperature and controller/IP output in response to changes in outside air temperature. Except between points A and B the controller/IP output is proportional to the discharge air temperature. Between points A and B, the coil discharge air temperature is the same as the outside air temperature. At points A and B, with a coil discharge temperature of 8 degrees C (47 degrees F), the output is 103 kPa (15 psig) and the valve is closed. At the fifth hour when the discharge temperature is at the 7 degrees C (45 degrees F) setpoint the controller/IP output is 62 kPa (9 psig) and the valve is half open. If the valve is closed at 8 degrees C (47 degrees F) outside air temperature and is half open with a 11 degrees C (20 degrees F) pick-up at -4 degrees C (25 degrees F) outside air temperature, then by extrapolation the valve is fully open at outside air temperatures of -16 degrees C (3 degrees F) and below. As soon as the air temperature leaving the coil, the incoming outside air temperature, and the heating action of the coil reach an equilibrium, the valve remains at a given position until the equilibrium is disturbed by changes in the outside air temperature.

(6) For a given load on the system, there is some optimal proportional sensitivity adjustment at which the controller will be in stable control and close to the setpoint. In some HVAC control applications, proportional control may function quite well with a high sensitivity adjustment. A high sensitivity adjustment

results in a narrow range of temperature drift from setpoint while the controller changes its output from full output to minimum output. Too high a sensitivity adjustment causes the control point to continuously overshoot and undershoot the setpoint. In other control applications, stable control may not be achievable with a high sensitivity adjustment. A low sensitivity adjustment results in a wide range of temperature drift from setpoint while the controller changes its output from full output to minimum output, but control is usually more stable. After the controller sensitivity has been adjusted, changing conditions of load due to seasonal and other factors tend to make the adjustment less than optimal. This phenomenon is a function of HVAC system capacity and HVAC system response to load changes. The sensitivity of a proportional controller to process variable changes is called proportional gain.

(7) As long as the proportional controller is controlling in a stable fashion, at varying load conditions, at some control point near the setpoint and the proportional gain setting is optimum, the controller has achieved the most precise control of which it is capable.

d. Proportional plus integral mode (PI).

(1) Many control applications require a controller that can eliminate offset due to load and can control very close to setpoint. To achieve closer control than is available from the proportional control mode, some automatic adjustment has to be made to the controller output to change the actuator position without changing the setpoint or the proportional gain setting. The method used to adjust the controller output for changes in load is called "integral mode". Integral mode adds a gain component algebraically to the controller output. This component is time proportional to the difference between the setpoint and the stable control point produced by the controller's proportional gain. This difference is caused by the offset due to load and is called steady-state error, which is the error between control point and setpoint when a balance between the load on the system versus the system capacity output and controller output is established. Steady-state error differs from the transitory error between setpoint and control point due to an upset in the process, such as a changing load or a step change in setpoint.

(2) The integral mode adds a component of output to the output of the controller that is produced by the controller's proportional gain. The size of the component is determined by the integral gain multiplied by the error. As the error decreases, the size of the component integrated to the output signal also decreases, and becomes zero when the controller is controlling at the setpoint. This added component of output causes the valve actuator stroke position to change. The change in valve capacity resulting from the change in the actuator stroke position upsets equilibrium or prevents equilibrium from occurring until the control point reaches setpoint. This action causes two things to happen. The resulting temperature change due to the change in valve actuator stroke position causes the proportional gain to change its component of output signal at a magnitude proportional to the change in the temperature being sensed. While this is happening, the error changes because the control point is closer to setpoint, and this in turn causes the integral gain to make a change in the magnitude of its component of the output signal due to the changing value of the magnitude of the error between control point and setpoint. There will be no further change in the proportional-mode or integral-mode output components when the steady-state error is zero. Because of the combined action of both of these control modes, the controller can reduce the offset to zero, or nearly zero, and can establish a steady-state equilibrium of HVAC system control at a value very near setpoint. This type of control is called Proportional-Integral (PI) control.

(3) See figure 2-14 for an illustration of proportional plus integral modes. The upper graph of the figure is identical to the upper graph of figure 2-13. The lower and middle graphs provide a comparison of the proportional mode controller output and resulting coil air discharge temperatures (light lines) versus those of a PI-mode controller (dark lines). In the beginning of the middle graph the integral mode component is positive and is added to the proportional mode component. This additional pressure closes the normally-open valve when the outside air temperature reaches setpoint 7 degrees C (45 degrees F), instead of at point A, at which the coil discharge air temperature is 7 degrees C (45 degrees F) plus half the controller throttling range or 8 degrees C (7 + 2/2) (47 degrees F (45 + 4/2)). The 8 degrees C (47 degrees

F) temperature would have occurred with proportional control, at point A. Similarly, when the outside air temperature falls to 8 degrees C (47 degrees F), the integral mode component prevents the valve from opening until the outside air temperature falls below 7 degrees C (45 degrees F). The proportional only mode controller would have begun to open the valve at 8 degrees C (47 degrees F) outside air temperature at point B. From the time the valve begins to open at the outside air temperature of 7 degrees C (45 degrees F) until the fifth hour when the outside air temperature is -4 degrees C (25 degrees F) the integral mode component of the controller output signal becomes smaller until at the fifth hour the integral mode component becomes zero. This additional pressure on the normally-open valve allows the coil air discharge temperature to remain at the 7 degrees C (45 degrees F) setpoint instead of controlling between 8 and 7 degrees C (47 and 45 degrees F) when the outside air temperature is between 7 and -4 degrees C (45 and 25 degrees F) as would have occurred with proportional only control. Similarly, when the outside air temperature falls below -4 degrees C (25 degrees F), the integral mode component is subtracted from the proportional mode component of the controller output signal to open the valve enough to keep the discharge air temperature at 7 degrees C (45 degrees F) rather than at 6 degrees C (43 degrees F) as would have occurred with proportional mode control.

Figure 2-14. Proportional plus integral control mode.

e. Effects of rapid load changes. The rate of change of load imposed on the HVAC system by the process affects how well the controller will perform its task of controlling at setpoint. The temperature of the outside air changes relatively slowly, and the temperature conditions inside also require some time to change. Inside conditions change as a function of air temperature changes made by the HVAC system, which warm up or cool down masses of material within the building. Inside conditions also change as functions of lighting load and occupancy. Because of these relatively slow rates of change, most of the HVAC processes that require gradual controller output changes can be controlled quite well with proportional plus integral (PI) control modes. Except for lighting loads on the HVAC system, these variable changes are relatively slow compared to the rates of change of variables that affect some non-HVAC processes. Lighting loads are sometimes imposed on the HVAC system quickly. This is an example of a step change in the process variable. The combined actions of proportional and integral modes are not always adequate to control rapidly changing variables.

f. Proportional-integral-derivative (PID) mode.

(1) Some processes require a controller that can respond to rapidly changing process variables. One answer to control of rapidly changing processes has been the addition of another control mode called derivative mode. When this control mode is added to proportional-integral control, the combination is known as proportional-integral-derivative (PID) control mode. The PID control mode adds a component algebraically to the output signal; this component is proportional to the rate of change of the error between the control point and setpoint. This automatic adjustment also affects the proportional and integral output components in a manner analogous to the way in which the integral component affects the proportional component. As the valve actuator stroke position changes, the temperature changes as a result of changing flow through the valve, and the rate of error signal change between the control point and setpoint varies as the control point comes closer to setpoint.

(2) There are a few HVAC control applications that are difficult for either P-mode or PI-mode control because of the fast rates of change of the process variables. One such application is the control of tankless heating converters, such as might be found in some domestic hot water heating applications. The I-mode component of PI-mode takes care of the varying range of offsets due to loads that occur in domestic hot water heating applications. For example, high-rise residential buildings have morning and evening peak periods of demand for hot water use. These peak demand periods drive the domestic hot water temperature to the low end of the offset range. Periods of relative nonoccupancy, such as late morning and early afternoon, drive the temperature to the high end of the offset range due to the minimal demand for domestic hot water use. Periods of relative inactivity during occupancy, such as late evening

and very early morning, require practically no domestic hot water. It is this period that defines the top end of the offset temperature range. The P-mode control alone does not control the water temperature very close to the controller setpoint in this kind of application. The addition of the I mode to the P mode makes the offset range much narrower than would occur with P mode alone. However, PI modes alone cannot handle the unpredictable diversity of demand as the peak periods start and end. What happens during the period of light demand for hot water use is that the turn-on of a shower or the startup of a dishwasher produces an upset in equilibrium that has a greater effect than the same event would have during a period of heavy demand. Periods of heavy demand tend to filter out some of the effect of a single turn-on.

(3) In the control loop applications where manual tuning is prescribed, the proportional mode constant is set as the result of a calculation. These applications cannot use the integral or derivative control modes. Self tuning is prescribed for these applications because finding the optimum settings manually is difficult and time consuming. When controllers self-tune, these settings are automatically optimized, and an optimized derivative-mode setting is selected due to the controller's self-tune feature. The effects of adding the I and D modes to the P mode is illustrated in figure 2-15, which shows the results that would be expected with a step change in setpoint. Step changes in setpoints rarely occur in HVAC control system applications. The illustration of the step change in setpoint is used to graphically explain the actions of P, I, and D modes.

Figure 2-15. Comparison of P mode, PI mode and PID mode for a step change in setpoint, and the contributions of each mode to controller output signal.

9. STANDARD HVAC SYSTEM CONTROL LOOPS.

a. Standard HVAC system control loops will be used for the design of HVAC control systems. The standard HVAC system control loops use the single-loop digital controller and additional components. These components are collectively called the control loop logic. The logic varies with the loop requirements, and its purposes are to interface the loops with the operational mode signals, to modify signals, and to interface with EMCS. The control loop logic is implemented by the use of combinations of relays and function modules.

b. Relay coils are activated by occupied-unoccupied, ventilation-delay, safety shutdown, EMCS-override, and other signals external to the HVAC control system. The contacts of the relays interrupt analog signals of controllers and function modules, provide inputs to on/off control loops (such as starter circuits), and operate HVAC control panel pilot lights.

c. All the relays, contacts, and function modules are defined for each loop. The relative physical locations of the relays and function modules will be assigned in the HVAC control panels.

d. Each of the standard HVAC control loops is described in this Technical Manual as the designer will show them on the contract drawings. Each control loop on the drawings will show the elements of the control loop, such as sensor/transmitters, controller, function modules, relay contacts, current to pneumatic signal converter (IP), and final actuator. The control loops will show all the field-mounted and panel-mounted devices for a standard loop. Each control loop will show function modules and relay contacts that are defined for the interfacing of analog control loops with on/off control loops.

10. SIZING AND SELECTION OF CONTROL SYSTEM DEVICES.

a. The designer will estimate the required motor horsepower of the HVAC control system air compressor, to provide the proper requirements for incorporation into the electrical power design.

b. Control system devices such as dampers and valves are sized on the basis of capacity requirements and allowable pressure drops and velocity ranges. The selection of the type of valve or damper is based on factors such as allowable leakage rates and available or practical size ranges.

c. The designer will analyze piping circuit pressure drops and their effects on the system's pressure drop due to the control valves, based on criteria presented in this manual. The designer will calculate the required liquid-flow coefficients (K_v (C_v)) for each valve and will show the K_v (C_v) for each valve. The designer will make sure that pumps are selected to include the pressure drop through the total circuit, and will then show the pressure against which the valve must close.

d. The designer will size control dampers based on the criteria presented in this manual and will show the size of each damper.

e. Oversizing of control devices would result in systems in which it would be difficult or impossible to obtain satisfactory control loop operation, regardless of the quality of the controller and components used. The designer must not assume that self-tuning controllers will compensate for oversizing of control devices. The I and D modes compensate for HVAC system load variations and HVAC system equilibrium upsets, but do not compensate for incorrect valve and damper sizing.

11. SIZING OF THE AIR COMPRESSOR MOTOR.

a. Calculation. The designer will estimate the required air compressor motor size, using equation 2-2, in order to coordinate with the power circuit serving the air compressor.

$$T_r = \frac{Q_c}{Q_d} \; x\,100 \qquad \text{(eq. 2-2)}$$

Where:

T_r = Air compressor running time (percent).
Q_c = Control system air consumption in standard milliliters per second (cubic inches per minute (scim)).
Q_d = Air compressor delivery in standard milliliters per second (cubic inches per minute (scim)).

b. Running-time criteria. The designer will use the following running-time criteria for sizing air compressor motors:

(1) New air compressors. The running-time design criterion for new air compressors is that the running time will not exceed 33-1/3 percent. This requires that, initially, the delivery capacity of the air compressor must be at least three times the estimated air consumption for the whole control system that it serves. The designer will count the number of each type of air-consuming device shown on the schematics and apply their characteristic consumption values to arrive at the consumption total. The designer will base the calculation on the air consumption for each device as shown in table 2-2.

(2) Existing air compressors. Control systems can be added to an existing air compressor until the running time reaches 50 percent. Exceeding 50 percent run-time risks excessive oil carryover into the air supply.

Table 2-2. Air consumptionof control devices.

Device	Air Consumption (ml/s (scim))

IP	8 (30)
Damper or valve positive positioner	6 (20)
EP	0 (0)

c. Application of air consumption values to sizing of air compressor motors. An example of the application of the values in table 2-2 is as follows:

5 IPs @ 8 ml/s (30 scim) each = 40 ml/s (150 scim)

15 valve and damper positioners @ 6 ml/s (20 scim) each = 90 ml/s (300 scim)

Total = 40 + 90 ml/s (150 + 300 scim) = 130 ml/s (450 scim)

Therefore, 130 ml/s (450 scim) x 3 (for 1/3 running time) = 390 ml/s (1,350 scim)

A manufacturer's catalog indicates that a typical ¼ hp compressor delivers about 390 ml/s (1,420 scim) and that a ½ hp compressor delivers about 712 ml/s (2,600 scim). The ¼ hp compressor would run 35 percent of the time, while a ½ hp compressor would run only 26 percent of the time. On the basis of this calculation, a power circuit capable of serving a ½ hp motor would provide adequate margin to compensate for variances in the air consumption of control devices selected by the control system vendor.

12. DETERMINATION OF CONTROL VALVE FLOW COEFFICIENT (K_v / C_v).

a. The control valve flow coefficient (K_v (C_v)) is a number representing the quantity of water, at a given temperature, that will flow through a valve with a given pressure drop. The designer will calculate and select the flow coefficient (K_v (C_v)) of all modulating control valves required for the design, and show the pressure against which the valve must close. The selection of the valve's K_v (C_v) provides the guidelines for the vendor's selection of the valve's port size. The close-off pressure information gives the required criterion for sizing the actuators for the valve selected by the vendor.

b. It should be noted that there is not yet an agreed upon international definition of control valve flow coefficients expressed in SI units (K_v). K_v data seems to be commonly expressed in units of either 1) m^3/hr @ Δp = 100 kPa (1 bar), or 2) L/s @ Δp = 1 kPa. Because of this, one needs to note the units associated with published K_v data and explicitly cite units associated with K_v when producing specifications or drawings. In this document K_v will be associated with the units of m^3/hr @ Δp = 100 kPa. (For conversion purposes: K_v (L/s @ 1 kPa) = 36 x K_v (m^3/hr @ 100 kPa).) A control valve flow coefficient expressed in inch-pound (I-P) units is denoted as C_v and is generally expressed with units of gallons per minute (gpm), at 60 degrees F, at a pressure drop of 1 psid.

c. A valve that is operated in a 2-position open and closed manner will be a line-size valve with the largest available K_v (C_v), in order to reduce the pressure drop across the valve and the pumping horsepower required. This applies to dual-temperature system changeover valves and dual-temperature fan coil unit 3-way valves.

d. The designer will select the K_v (C_v) of the valve based on the maximum flow and the pressure drop for the valve. The selected control valve will be checked against manufacturer's catalogs to insure that such a valve K_v (C_v) is available in a product of the control valve manufacturers.

e. To insure good control characteristics, the pressure drop through the control valve at full flow must be greater than the pressure drop through the coil and piping circuit (without the control valve) between the point where the piping circuit connects to the supply and return mains. The pressure drop through the control valve will then be at least 50 percent of the total pressure drop through the circuit. The valve K_v (C_v) must be specified accordingly. For liquid service, the vendor may supply a control valve with a plus 25 percent deviation from that specified. The designer must check to insure that the pressure drop through the valve will be acceptable at both the specified K_v (C_v) and at a K_v (C_v) 25 percent greater than specified. As the K_v (C_v) increases the pressure drop through the valve decreases. For steam service, the vendor must supply a control valve with a K_v (C_v) not less than the value specified and not larger than the manufacturer's next larger value.

13. CALCULATION OF LIQUID CONTROL VALVE FLOW COEFFICIENT (K_v / C_v).

a. A physical phenomenon known as cavitation is a cause of valve failure. Cavitation is caused when the velocity through the valve creates an absolute pressure lower than the vaporization pressure of the liquid. To avoid cavitation, use equation 2-3 to determine the maximum allowable pressure drop through open valves.

$$\Delta p_m = k_m (p_e - p_v) \qquad \text{(eq. 2-3)}$$

Where:

Δp_m = maximum allowable pressure drop through the valve, kPa (psid)

k_m = valve pressure recovery coefficient (use 0.45)

p_e = absolute pressure entering the valve, kPa (psia) (by design calculations).

p_v = absolute vapor pressure of the liquid, kPa (psia) (from steam tables).

For example: If 111 degree C (200 degree F) water with an equivalent vapor pressure of 79.5 kPa (11.53 psia) is to flow through a valve with an inlet pressure of 138 kPa (239 kPa (abs.)) (20 psig (34.7 psia)), the maximum allowable pressure drop through the valve will be, as a result of using equation 2-3, 71.9 kPa (10.43 psid).

b. The pressure drop through the control valve must be at least 50 percent of the total pressure drop, at full flow, through the circuit. Continuing with the above example, assume a coil with a flow of 11.35 m^3/hr (50 gpm) and a 27.6 kPa (4 psig) drop through the coil, piping and fittings between the mains. The maximum allowable pressure drop to avoid cavitation was calculated to be 71.9 kPa (10.43 psid). An initial valve pressure drop selection of 41.4 kPa (6 psid) will prevent cavitation and provide sufficient valve pressure drop for good control. The K_v (C_v) can be calculated from equation 2-4:

$$K_v = \frac{Q}{\sqrt{\frac{\Delta p}{G}}} \qquad \text{(eq. 2-4 SI)}$$

Where:

K_v = flow coefficient, m^3/hr @ Δp = 100 kPa (1 bar)

Q = flow, m^3/hr

Δp = pressure drop through valve, in bars (100 kPa = 1 bar)

G = specific gravity of the liquid.

$$C_v = \frac{Q}{\sqrt{\frac{\Delta p}{G}}} \qquad \text{(eq. 2-4 IP)}$$

Where:

C_v = flow coefficient, gpm @ Δp = 1 psi

Q = flow, gpm

Δp = pressure drop through valve, psi

G = specific gravity of the liquid.

The specific gravity of water is 1. As the result of using equation 2-4, K_v = 17.64 (C_v = 20.41). If the designer specifies a K_v of 17 (C_v of 20), the vendor can provide a valve with a K_v in the range of 17 to 21.25 (C_v in the range of 20 to 25). If the vendor provides a valve with a K_v of 21.25 (C_v of 25), the pressure drop would be 28.5 kPa (4 psid) as calculated by using equation 2-4. This valve selection would make the pressure drop through the valve 50 percent of the drop through the total piping circuit. Knowing that a plus 25 percent deviation in valve K_v (C_v) selection is allowable, the designer could select a K_v (C_v) which would result in a larger pressure drop through the control valve. With a K_v of 14 (C_v of 16), the valve pressure drop will be 65.7 kPa (9.76 psid) as calculated by the use of equation 2-4. This is less than the maximum allowable valve pressure drop of 71.9 kPa (10.43 psid), and the total circuit pressure drop would be 27.6 kPa (4 psid) (coil and piping) plus 65.7 kPa (9.76 psid) (valve) = 93.3 kPa (13.76 psid). A K_v of 14 (C_v of 16) would result in the pressure through the valve being 71 percent of the drop through the piping circuit. If the designer specifies a K_v of 14 (C_v of 16), the vendor can provide a valve with a K_v in the range of 14 to 17.5 (C_v in the range of 16 to 20). Cavitation will not be a problem in this example if the water temperature used is 111 degrees C (200 degrees F), or less. In most HVAC control valve applications, water temperatures are less than 111 degrees C (200 degrees F). When the hot water temperature is scheduled from outside air, 111 degree C (200 degree F) water will occur only on design heating days. Also, cavitation should not be a problem in hydronic systems when expansion tank pressures are adjusted so that cavitation is not present at the pump inlet. Selection by the designer of a control valve K_v of 17 (C_v of 20) rather than a K_v of 14 (C_v of 16) would be technically acceptable according to the criteria of this EI, but would not be a good engineering practice. If the larger K_v (C_v) is used as the basis of design, the contractor may provide a valve with a K_v of 21.25 (C_v of 25). A K_v of 21.25 (C_v of 25) may result in the valve pressure drop being less than 50 percent of the drop through the circuit. The basis of design valve K_v (C_v) is selected because of the drop expected through the basis of design coil and the piping system configuration shown by the designer. The coil provided by the contractor, field piping conditions and aging of the piping system are factors which could result in the installed valve being less than 50 percent of the drop through the coil when the contractor provides a valve with the largest K_v (C_v) permissible.

c. Review of manufacturers' catalogs shows the ready availability of valves with K_v in the range of 14 to 22 (C_v in the range of 16 to 25), as listed in table 2-3.

Table 2-3. Available control valves with K_v (C_v) in the range of 14 to 22 (16 to 25)

Valve Type	Valve Size	K_v (m^3/hr @ Δp = 100 kPa) / C_v (gpm @ Δp = 1 psi)
Normally-open water valve	25 mm (1-inch)	15.5 / 18
Normally-open water valve	32 mm (1-1/4 inch)	19.2 / 22.2
Normally-open water valve	32 mm (1-1/4 inch)	14.7 / 17
Normally-open water valve	32 mm (1-1/4 inch)	13.8 / 16
Normally-open water valve	40 mm (1-1/2 inch)	21.6 / 25
Normally-open water valve	40 mm (1-1/2 inch)	17.3 / 20
Three-way valve	25 mm (1-inch)	15.5 / 18
Three-way valve	25 mm (1-inch)	15.6 / 18.1
Three-way valve	32 mm (1-1/4 inch)	13.8 / 16
Three-way valve	32 mm (1-1/4 inch)	14.7 / 17
Three-way valve	40 mm (1-1/2 inch)	21.6 / 25
Three-way valve	40 mm (1-1/2 inch)	18.2 / 21

At this point, the designer will check the line size of the piping circuit and check the piping size of the reducing fittings. A good choice for the designer would be to show the K_v at 14 (C_v at 16) and the maximum close-off pressure at 140 kPa (20 psig) in this example because there are more control valves available in the range of 14 to 17.5 (16 to 20) (a plus 25 percent deviation) than there are in the range of 17.5 to 21.25 (20 to 25) (a plus 25 percent deviation).

c. The K_v (C_v) calculations for selecting a butterfly valve are identical to those for any valve except that the valve K_v (C_v) is selected using the calculated K_v (C_v) at a maximum of 70 degrees open when the valve is to be used in a modulating control application. (The butterfly valve does not have a good modulating control characteristic curve between 70 degrees and 90 degrees open.)

d. When selecting control valves for liquid service other than water, the designer will take into account the specific gravity of the liquid. If the liquid has a specific gravity of 1.05, for the same flow rate, the liquid flow will produce a 5 percent greater pressure drop through the valve.

14. CALCULATION OF STEAM CONTROL VALVE FLOW COEFFICIENT (K_v / C_v).

a. Calculating and selecting the required K_v (C_v) of a steam valve requires the designer to consider:

(1) The saturated or superheated condition of the steam.
(2) The inlet pressure at the valve.
(3) The minimum required steam pressure entering the steam-condensing apparatus at peak heating load.

b. The designer will use equations that correct the liquid K_v (C_v) for the compressibility of the steam. The designer will take into account in his calculations that the limiting factor of steam flow through a control valve is the flow that occurs at critical pressure drop, and will use an appropriate factor for calculating the critical pressure drop. The designer will calculate the critical pressure drop across the valve by using equation 2-5.

$$\Delta p_{cr} = k_m \; x \; p_e \qquad \text{(eq. 2-5)}$$

Where:

Δp_{cr} = critical pressure drop, kPa (psid)

k_m = valve pressure recovery coefficient, (use 0.45)

p_e = Absolute pressure of the steam entering the valve, kPa (psia)

c. For steam control valve application using saturated steam at a pressure drop across the valve less than the critical pressure drop, the K_v (C_v) will be calculated using equation 2-6.

$$K_v = \frac{Q_s}{0.161\sqrt{(p_e + p_o)\Delta p}} \qquad \text{(eq. 2-6 SI)}$$

Where:

K_v = flow coefficient, m^3/hr @ Δp = 100 kPa (1 bar)

Q_s = steam flow, kg/hr

0.161 = compressibility and conversion factor.

p_e = absolute steam pressure entering the valve, kPa

p_o = absolute pressure entering steam coil, kPa

Δp = pressure drop through valve, kPa

$$C_v = \frac{Q_s}{2.11\sqrt{(p_e + p_o)\Delta p}} \qquad \text{(eq. 2-6 IP)}$$

Where:

C_v = flow coefficient, gpm @ Δp = 1 psi

Q_s = steam flow, lb/hr

2.11 = compressibility and conversion factor.

p_e = absolute steam pressure entering the valve, psia

p_o = absolute pressure entering steam coil, psia

Δp = pressure drop through valve, psid

d. When the drop across the valve will be at or greater than the critical pressure drop and the steam is saturated, the K_v (C_v) will be calculated using equation 2-7.

$$K_v = \frac{Q_s}{0.133\ p_e} \qquad \text{(eq 2-7 SI)}$$

Where:

K_v = flow coefficient, m^3/hr @ Δp = 100 kPa (1 bar)

Q_s = steam flow, kg/hr

0.133 = compressibility and conversion factor.

p_e = absolute pressure of entering steam, kPa

$$C_v = \frac{Q_s}{1.74\ p_e} \qquad \text{(eq 2-7 IP)}$$

Where:

C_v = flow coefficient, gpm @ Δp = 1 psid

Q_s = steam flow, lb/hr

1.74 = compressibility and conversion factor.

p_e = absolute pressure of entering steam, psia

e. For superheated steam applications where the drop across the valve is less than the critical pressure, the K_v (C_v) will be calculated using equation 2-8.

$$K_v = \frac{Q_s\,(1 + 0.00131\,(T_{sh}\,)\,)}{0.161\sqrt{(\,p_e + p_o\,)\,\Delta\,p}} \qquad \text{(eq. 2-8 SI)}$$

Where:

K_v = flow coefficient, m^3/hr @ Δp = 100 kPa (1 bar)

Q_s = steam flow, kg/hr

(1 + 0.00131 (T_{sh})) = superheat factor.

T_{sh} = superheat temperature, degrees C

0.161 = compressibility and conversion factor.

p_e = absolute pressure of the steam entering the valve, kPa

p_o = absolute pressure of the steam entering the coil, kPa

Δp = pressure drop through the valve, kPa

$$C_v = \frac{Q_s\,(1 + 0.0007\,(T_{sh}\,)\,)}{2.11\sqrt{(\,p_e + p_o\,)\,\Delta\,p}} \qquad \text{(eq. 2-8 IP)}$$

Where:

C_v = flow coefficient, gpm @ Δp = 1 psid

Q_s = steam flow, lb/hr

(1 + 0.0007 (T_{sh})) = superheat factor.

T_{sh} = superheat temperature, degrees F

2.11 = compressibility and conversion factor.

p_e = absolute pressure of the steam entering the valve, psia

p_o = absolute pressure of the steam entering the coil, psia

Δp = pressure drop through the valve, psid

T_{sh} is the superheat temperature of the steam reduced to the lower pressure from the higher pressure.

f. For superheated steam applications with the pressure drop across the valve at or greater than the valve's critical pressure, the K_v (C_v) will be calculated using equation 2-9.

$$K_v = \frac{Q_s(1 + 0.00131(T_{sh}))}{0.133\ p_e} \qquad \text{(eq. 2-9 SI)}$$

Where:

K_v = flow coefficient, m^3/hr @ Δp = 100 kPa (1 bar)

Q_s = steam flow, kg/hr

(1 + 0.00131 (T_{sh})) = superheat factor

T_{sh} = superheat temperature, degrees C

0.133 = compressibility and conversion factor.

p_e = absolute pressure of the steam entering the valve, kPa

$$C_v = \frac{Q_s(1 + 0.0007(T_{sh}))}{1.74\ p_e} \qquad \text{(eq. 2-9 IP)}$$

Where:

C_v = flow coefficient, gpm @ Δp = 1 psid

Q_s = steam flow, lb/hr

(1 + 0.0007 (T_{sh})) = superheat factor

T_{sh} = superheat temperature, degrees F

1.74 = compressibility and conversion factor.

p_e = absolute pressure of the steam entering the valve, psia

g. An example of the calculation required for the K_v (C_v) of a valve designed to handle saturated steam when the pressure drop will be less than critical is as follows: An air coil requires 113.4 kg/hr (250 lb/hr) of steam entering the coil at 13.8 kPa (2 psig). The steam is generated by a local boiler with the boiler's operating pressure switches set at 34.5 kPa (5 psig) "on" and 55.2 kPa (8 psig) "off". Assuming 34.5 kPa (5 psig) at peak load at the valve inlet and 13.8 kPa (2 psig) at the valve outlet, the K_v = 9.8 (C_v = 11.34) as a result of using equation 2-6. In this example, critical pressure drop is not a limiting factor, because the coil pressure at critical pressure drop would be (101.4 + 34.5) x .45 = 61.2 kPa ((14.7 + 5) x 0.45 = 8.87 psia), which is not possible unless the system is designed to operate at a high vacuum. A check of manufacturers' literature shows K_v of 8.6, 10.4, 11.2 (C_v of 10, 12, and 13) in a 25 mm (1 inch) valve. A valve K_v of 10.4 (C_v of 12) would be a good design choice for this application.

h. When the calculation for steam valve selection results in a K_v greater than 47.5 (C_v greater than 55), the designer will select two valves, sized to split the flow unevenly between the valves; a good ratio is one-third of the flow for the smaller valve and two-thirds of the flow for the larger valve, which, however, is not always achievable from the normally available stock of valves. As an example of the calculation required, a steam converter requires 2,722 kg/hr (6,000 lb/hr) of steam at 7 kPa (1 psig) entering the converter shell at peak load with the steam pressure reduced from 862 kPa (125 psig) to 172 kPa (25 psig) near the convertor. The critical pressure drop is 123 kPa (17.87 psid) as a result of using equation 2-5, and the pressure drop across the valve is 172 - 7 = 165 kPa (25-1 = 24 psid), which is greater than the critical pressure drop. The constant-enthalpy process of reducing the steam pressure from 862 kPa (125 psig) to 172 kPa (25 psig) results in 26.7 degrees C (48 degrees F) of superheat. The required K_v (C_v) is 77.5 (89.1) as a result of using equation 2-9. This K_v (C_v) is 3 percent larger than it would have been if the steam were saturated at 172 kPa (25 psig) and the K_v (C_v) were calculated from equation 2.7. In this example, the designer should show two valves in parallel and attempt to find available valves with K_vs of approximately 52 and 26 (C_vs of approximately 60 and 30). A check of manufacturers' catalogs for this example results in the designer showing valves with K_vs of 60.5 and 19.9 (C_vs of 70 and 23) and a close-off pressure of 172 kPa (25 psig). The selection results from the following available control valve combinations, any of which will meet the design intent:

(1) one 65 mm (2-1/2 inch) valve, K_v = 64 (C_v = 74), and one 40 mm (1-1/2 inch) valve, K_v = 15.6 (C_v = 18), or total K_v = 79.6 (C_v = 92)

(2) one 65 mm (2-1/2 inch) valve, K_v = 60.5 (C_v = 70), and one 40 mm (1-1/2 inch) valve, K_v = 19.9 (C_v = 23), or total K_v = 80.4 (C_v = 93)

(3) one 80 mm (3-inch) valve, K_v = 61.1 (C_v = 70.7), and one 40 mm (1-1/2 inch) valve, K_v = 20 (C_v = 23.1), or total K_v = 81.1 (C_v = 93.8)

15. DETERMINING VALVE ACTUATOR CLOSE-OFF PRESSURE RATINGS.

a. The close-off rating indicates the pressure against which a valve must be able to close. Figure 2-16 is a normally open pneumatic valve. F1 is the force of the air pressure acting to close the valve. F2 is the opposing spring pressure and F3 is the force exerted by the fluid (water pump head or steam pressure). The value of F3 is actually the difference in pressure on the two sides of the valve plug. The valve and actuator must be able to close the valve against fluid force F3. The expected value of F3 is part of the hydraulic design of the piping system and depends upon the location of the valve within the system. To provide for a safety factor, the pressure the valve must close against is normally specified at several times the value actually expected at the valve's location. A three-way valve must operate against a similar fluid pressure in moving from one position to another.

Figure 2-16. Typical normally-open pneumatic valve.

b. In piping circuits with two-way valves, the flow is not constant. As the valve closes, the flow decreases. As the flow decreases, the friction losses decrease and the pump delivers more pressure. As a valve shuts off (worst condition) the valve must take the full pump pressure at zero flow (dead head). To provide a measure of safety and to allow for differences between design and installed conditions, close-off pressure ratings for two-way valves should specified to be 100% to 125% of the rated pump pressure at the design flow rate.

Figure 2-17. Close-off pressure for two-way valves.

c. In piping circuits with three-way valves, the flow is constant. Therefore, friction losses are constant and the pump pressure is constant. The three-way valve must close against the pressure difference between the supply and return headers. Three-way valves should be sized for ß = 0.7 and the close-off pressure rating should be specified to be twice the valve pressure drop.

Figure 2-18. Close-off pressure for three-way valves.

16. SURGE PROTECTION PROVISIONS FOR TRANSMITTER AND CONTROL WIRING. Because HVAC control system transmitter signals and single-loop controller signals are interfaced with EMCS, the appropriate surge protection will be provided in the HVAC control system design as described in TM 5-815-2.